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SECTION 6

PRIME MOVERS

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6.1 STEAM PRIME MOVERS

6.1.1 Steam Engines and Steam Turbines

Steam prime movers are either reciprocating engines or turbines, the former being the older, dominant type until 1900. Reciprocating engines offer low speed (100 to 400 r/min), high efficiency in small sizes (less than 500 hp), and high starting torque. In the Industrial Revolution, they powered

mills and steam locomotives. Steam turbines are a product of the twentieth century and have established a wide usefulness as prime movers. They completely dominate the field of power generation and are a major prime mover for variable-speed applications in ship propulsion (through gears), centrifugal pumps, compressors, and blowers. Single steam turbines can be built in greater capacities (over 1,000,000 kW) than any other prime mover. Turbines offer high speeds (1800 to 25,000 r/min) and high efficiencies (over 85% in larger units); require minimum floor space with relatively low weight; need no internal lubrication; and operate at high steam pressures (5000 lb/in² [gage]), high steam temperatures (1050°F), and low vacuums (0.5 inHg [abs]). Steam turbines have no reciprocating mass (with resulting vibrations) nor parts subject to friction wear (except bearings), and consequently provide very high reliability at low maintenance costs.

6.1.2 Steam-Engine Types and Application

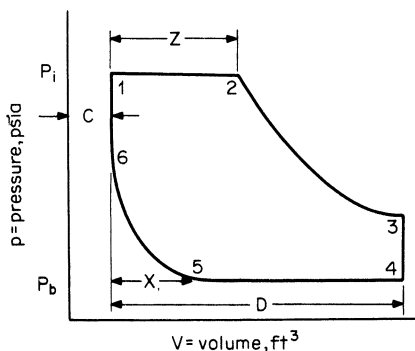
The former great diversity in engine types has been reduced so that (1) simple D-slide engines (less than 0.100 hp) are used for auxiliary drive and (2) single-cylinder counterflow and uniflow engines (less than 1000 hp), with Corliss or poppet-type valve gear, are used for generator or equipment drive in factories, office buildings, paper mills, hospitals, laundries, and process applications (where non-condensing by-product power operations prevail). Multiple-expansion, multicylinder constructions are largely obsolete except for some marine applications. Although engines as large as 7500 kW have been built and are still found in service, the field is generally limited to engines less than 500 kW in size. Engine governing is by flyball or flywheel types to (1) throttle steam supply or (2) vary cutoff.

6.1.3 Steam-Engine Performance

The basic thermodynamic cycle is shown in Fig. 6-1. The net work of the cycle is represented by the area enclosed within the diagram and is represented by the mean effective pressure (mep), that is, the net work (area) divided by the length of the diagram. The power output is computed by the "plan" equation:

$$\text{hp} = \frac{p_m L a n}{33,000} \quad (6-1)$$

where hp = horsepower; p_m = mep, pounds per square inch; L = length of stroke, feet; a = net piston area, square inches; and n = number of cycles completed per minute.



Where	Usual value
D = displacement	0.05–20 ft ³
C = clearance	0.03–0.2
Z = cutoff, fraction of D	0.1–0.6
X = compression, fraction of D	0.1–0.8
p_i = initial pressure	100–300 lb/in ² (abs.)
p_b = back pressure	2–30 lb/in ² (abs.)
p_m = mean effective pressure	50–125 lb/in ² (abs.)

FIGURE 6-1 Pressure-volume diagram for a steam-engine cycle. Phase 1-2, constant-pressure admission at P_i ; phase 2-3, expansion, $p_v = C$; phase 3-4, release; phase 4-5, constant-pressure exhaust pipe at P_b ; phase 5-6, compression, $p_v = C$; phase 6-1, constant-volume admission.

The theoretical mep and horsepower are larger than the actual indicated values and are customarily related by a diagram factor ranging between 0.5 and 0.95. The shaft or brake mep and horsepower are lower still, with mechanical efficiency ranging between 0.8 and 0.95.

6.1.4 Steam Turbines—General

1. *Expansion of steam through nozzles and buckets.* Basically, steam turbines are a series of calibrated nozzles through which heat energy is converted into kinetic energy which, in turn, is transferred to wheels or drums and delivered at the end of a rotating shaft as usable power.

2. *Impulse, reaction, and Curtis staging.* Turbines are built in two distinct types: (1) impulse and (2) reaction. *Impulse turbines* have stationary nozzles, and the total stage pressure drop is taken across them. The kinetic energy generated is absorbed by the rotating buckets at essentially constant static pressure. Increased pressure drop can be efficiently utilized in a single stage (at constant wheel speed) by adding a row of turning vanes or “intermediates” which are followed by a second row of buckets. This is commonly called a *Curtis* or 2-row stage.

In the *reaction* design, both the stationary and rotating parts contain nozzles, and an approximately equal pressure drop is taken across each. The pressure drop across the rotating parts of reaction-design turbines requires full circumferential admission and much closer leakage control.

To illustrate the variations in energy-absorbing capacities of an impulse stage, a 2-row impulse stage, and a reaction stage, one must start with the general energy equation as applied to a nozzle:

$$\frac{V_1^2}{2gJ} + H_1 = \frac{V_2^2}{2gJ} + H_2 \quad (6-2)$$

which is reduced to

$$\text{Jet velocity, ft/s} = 223.7\sqrt{\Delta H} \quad (6-3)$$

where V_1 is assumed to be zero, and ΔH is the enthalpy drop (isentropic expansion) in Btu per pound as obtained from the Mollier chart for steam (Fig. 6-2).

Assuming a typical wheel pitch line speed (W) of 550 ft/s and initial steam conditions of 400 lb/in² (abs.), 700°F ($H_1 = 1363.4$ Btu/lb), the optimum energy-absorbing capacities of each type can be derived.

Table 6-1 illustrates that the energy-absorbing capability of the Curtis stage is 4 times that of an impulse stage and 8 times that of a reaction stage.

Because of this capability, the 2-row Curtis stage has found many applications in the process industries for small mechanical-drive use (up to 1000 hp) where the inlet steam can be taken from one process header and the exhaust steam sent out to a lower-pressure process header. As energy costs increase, however, the lower efficiency attainable with these small-volume-flow single-stage units offsets some of the desirable features (e.g., speed control, low cost, etc.). All modern turbines over 1000 hp are multistage for good efficiency, varying from 3 to 4 stages on noncondensing units with a small pressure ratio up to 20 or more stages on large reheat condensing units. Reaction (Parsons) designs generally have more stages than impulse (Rateau) designs. All large units have an impulse (1- or 2-row) first stage because there is no pressure drop on the moving rows, which makes it more suitable for partial-arc admission.

3. *The control stage.* The first stage of the turbine must be designed to pass the maximum flow through the unit at rated inlet steam conditions. The pressure required at less than rated flow will decrease if the nozzle area is held constant, resulting in a throttling loss through the control valves of the unit at partial flows. Very early in the development of steam turbines, it was recognized that if full throttle pressure could be made available to the first-stage nozzles across the load range, the maximum isentropic energy that would be available for work and overall efficiency would be increased at part load. Most first stages now use sectionalized first-stage nozzle plates with 4, 6, or 8 separate ports (depending on steam conditions, unit size, and manufacturer).

P-V-T Values for Dry Saturated Steam

Pressure, in. Hg abs	Temp., °F	Sp. vol., ft ³ /lb
0.5	58.8	1250
1.0	79.0	652
1.5	91.7	445
2.0	101.1	339
3.0	115.1	232

Pressure, lb/in ² (abs.)		
14.7	212.0	26.8
50	281.0	8.52
100	327.8	4.43
200	381.8	2.29
300	417.3	1.543
400	444.6	1.161
600	486.2	0.770
900	532.0	0.501
1200	567.2	0.362
1800	621.0	0.218
2400	662.0	0.141
3206	705.4	0.050

NOTE: 1 in = 25.4 mm; $t_c = (t_F - 32)/1.8$; 1 ft³/lb = 0.0624 m³/kg.

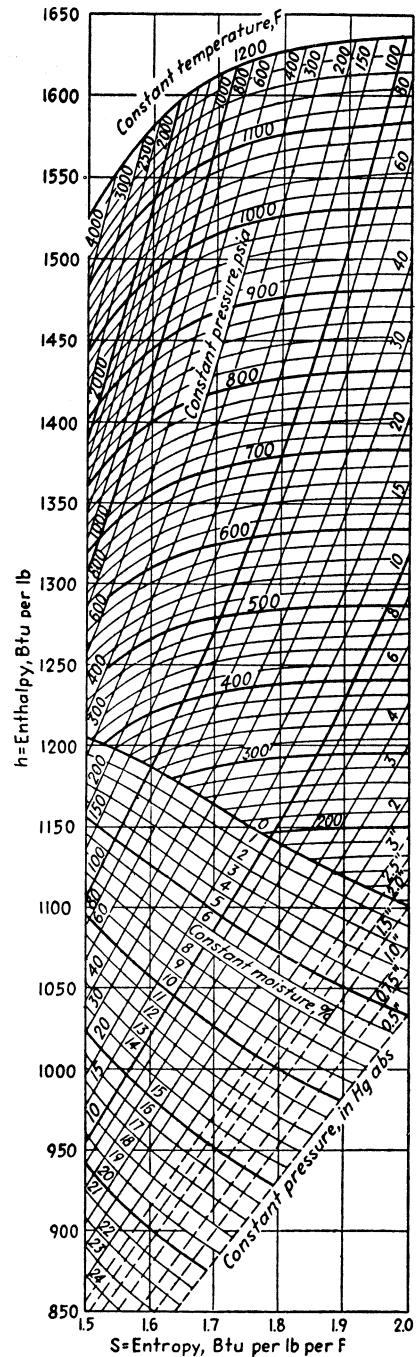


FIGURE 6-2 Mollier chart for steam (ASME steam tables.)

TABLE 6-1 Energy-Absorbing Capability of the Curtis Stage

Stage type	Impulse	2-row wheel	Reaction		Combined
			Sta.	Rot.	
Theoretical W/V for peak efficiency	0.50	0.25	...	...	0.707
Required jet velocity, ft/s	1100	2200	550	550	
ΔH required, Btu/lb	24.2	96.6	6.045	6.045	12.09
Required P_2 lb/in ² (abs) from Mollier chart	327	168	380.6	361.9	361.9
Stage P_1/P_2	1.224	2.38	1.052	1.105	1.105

Note: 1 ft/s = 0.3048 m/s; 1 lb/in² = 0.06895 bar; 1 Btu/lb = 2.326 kJ/kg.

The flow to each port is controlled by its own valve, and the valves are opened sequentially. As each valve is opened to its governing point, the full throttle pressure (minus stop-valve and control-valve pressure loss) becomes available to the arc of nozzles fed by that valve. The overall result is a greater availability of energy to do work.

4. *Steam-path design.* Condensing-turbine sizes increase with the development of longer last-stage buckets and, consequently, the last-stage dimensions (length and diameter) are the first to be determined; these dimensions fix the diameter of the L-1 stage and the optimum energy (pressure drop) which can be placed on that stage. This stage in turn defines the parameters of the L-2 stage and so on up to the first stage, and it can be said that steam paths of turbines are designed backward except for the first stage. In the 1970s, the largest-capacity single-flow condensing turbine was approximately 120,000 kW. Larger ratings are obtained by multiplying the number of exhaust stages (usually the last 5 to 7 stages are involved) by 2, 4, 6, or 8 times to satisfy the rating requirements. This practice is limited to the larger blades to round out a product line to well over 1,000,000 kW.

6.1.5 Turbine Efficiency

1. *Nozzle and bucket.* The turbine stage efficiency is defined as the actual energy delivered to the rotating blades divided by the ideal energy released to the stage in an isentropic expansion from P_1 to P_2 of the stage. The most important factors determining the stage efficiency are the relationship of the mean blade speed to the theoretical steam velocity, the aspect ratio (blade length/passage width), and the aerodynamic shape of the passages. Figure 6-3 describes the typical variation in nozzle and bucket efficiencies with velocity ratio and nozzle height.

2. Losses

Clearance leakage. A 100% efficiency cannot be obtained because of friction in the blading and clearance between the stationary and rotating parts, and because the nozzle angle cannot be zero degrees. Axial clearance increases in the stages further from the thrust bearing to satisfy the need to maintain a minimum clearance at extreme operating conditions when the differential expansion between the light rotor and heavy casing is at its worst. To reduce this leakage, radial spillbands are used. These thin, metal-strip seals may be attached to the diaphragm or casing and extend close to the shroud bands covering the rotating blades. This clearance can be kept quite close (0.020 to 0.060 in), and axial changes in the rotor position do not affect the clearance since the

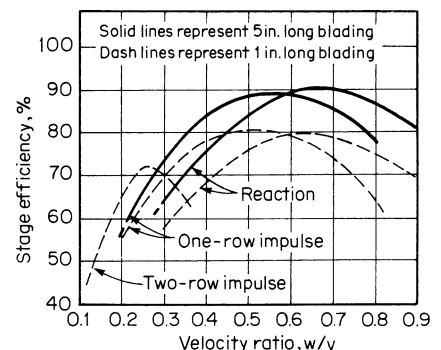


FIGURE 6-3 Approximate relative efficiencies of turbine stage types.

spillbands ride over the shrouds. The need to control the clearance leakage area is especially important on reaction stages with small blade heights because of the pressure drop across the moving blades.

Nozzle leakage. Leakage around the nozzles between the bore of the blade ring or nozzle diaphragm and the drum or rotor must be kept to a minimum. This leakage is controlled through the use of a metallic labyrinth packing which consists of a single ring with multiple teeth arranged to change the direction of the steam as well as to minimize the leakage area. Labyrinth packings are also used at the shaft ends to step the pressure down at the high-pressure end and to seal the shaft at the vacuum end.

Rotation loss. Rotation of the rotor consist of losses due to the rotation of the disks, the blades, and shrouds. Partial-arc impulse stages have a greater windage loss within the idle buckets. Rotation losses vary directly with the steam density, the fifth power of the pitch diameter, and the third power of the rpm. In general, the windage loss amounts to less than 1% of stage output at normal rated output. At no-load conditions, windage loss for noncondensing turbines approximates 1.5% of the rating per 100 lb/in² exhaust pressure, and on condensing units approximates from 0.4% to 1.0% of the rating at 1.5 inHg (abs) exhaust pressure.

Carryover loss. A carryover loss (about 3%) occurs on certain stages when the kinetic energy of the steam leaving the rotating blades cannot be recovered by the following stage because of a difference in stage diameters or a large axial space between adjacent stages. Typically, this happens in control stages and in the last stages of noncondensing sections. The last stages of condensing turbines have the largest carryover losses (normally referred to as exhaust loss) because of the large variations in exhaust volumetric flow with exhaust pressure and the large variation of stage pressure ratio with load. Stages preceding the last operate with essentially a constant pressure ratio down to very low loads and consequently can be designed for peak efficiency at a wide range of loads.

Leaving loss. Condensing turbines are frequently “frame sized” by last-stage blade height. It is sometimes economical to size the unit with exhaust loss equal to 5% deterioration in overall turbine performance at the design point (valves wide-open throttle flow and 1.5 inHg [abs] exhaust pressure) when the normal expected exhaust pressure will be higher or the unit will be operating at part load for a large part of the time.

Nozzle end loss, partial arc. Control stages and partial-arc impulse stages are subject to end losses at the interface of the active and inactive portions of the blading as the stagnant steam within the idle bucket passages enters the active arc of nozzles and must be accelerated. There is also a greater turbulence in the steam jet at both ends of the active arc. In partial-arc impulse stages, the increase in efficiency due to larger blade heights (aspect ratio) is partially offset by increased rotation and end losses, and there is an optimum to this proportioning beyond which there is an overall loss.

Supersaturation and moisture loss. Moisture in the steam causes supersaturation and moisture losses in the stage. The acceleration of the moisture particles is less than that of the steam, causing a momentum loss as the steam strikes the particles. The moisture particles enter the moving blades (buckets) at a negative velocity relative to the blades, resulting in a braking force on the back of the blades. Supersaturation is a temporary state of supercooling as the steam is rapidly expanded from a superheated state to the wet region before any condensation has begun. The density is greater than when in equilibrium, resulting in a lower velocity as the steam leaves the nozzle. As soon as some condensation occurs at approximately 3.5% moisture, according to Yellot, a state of equilibrium is almost instantly achieved and supersaturation ceases.

3. Turbine efficiency. The internal used energy of the stage is obtained by multiplying the isentropic energy available to the stage by the stage efficiency. The sum of the used energies of all stages in the turbine represents the total used energy of the turbine. The internal efficiency of the turbine can be obtained by dividing the total used energy by the overall isentropic available energy from throttle pressure and temperature conditions to the exhaust pressure. (Note: The sum of the available energies of the stages is greater than the overall available energy and represents the reheat factor or

gain attributable to the unused energy of preceding stages becoming available to following stages.) The use of overall available energy will automatically account for pressure-drop losses occurring in stop valves, control valves, exhaust hood, and piping between HP and LP elements. Other losses which must be accounted for to arrive at the turbine overall efficiency include valve-stem and shaft-end packing leakages and bearing and oil-pump losses. Determination of the overall efficiency of a turbine and its driven equipment must take into account the losses of gears or generators and their bearings as well.

6.1.6 Turbine Construction

Since the early 1900s, horizontal-shaft units have been universally used. Horizontal units may be single-shaft or double-shaft, with single, double, or triple steam cylinders on one shaft. These modern units may be throttle or multiple-nozzle governed, have one or more steam extraction points, and exhibit innumerable variations in construction. Figure 6-4 shows several of the more commonly used types of turbines in schematic cross sections.

Steam turbines may be classified into several broad categories, according to the basic purpose and design of the steam path: (1) straight condensing, (2) straight noncondensing, (3) uncontrolled extraction, (4) single, double, or triple controlled extraction, and (5) reheat. Various combinations of these features may be present in a typical unit, and occasionally unusual variations on the above types may be seen.

Figure 6-5 is a cross section of a modern automatic-extraction turbine, showing the details of construction. A steam turbine consists of the following basic parts: (1 to 3) steam path made up of rotating

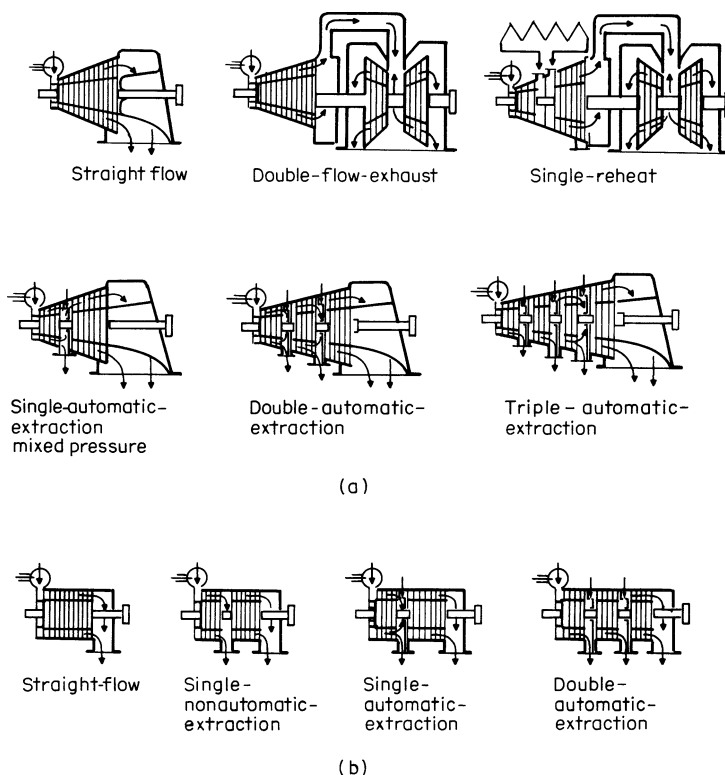


FIGURE 6-4 (a) Condensing turbines (exhaust at backpressures less than atmospheric); (b) noncondensing turbines (wide range of backpressures). (*General Electric.*)

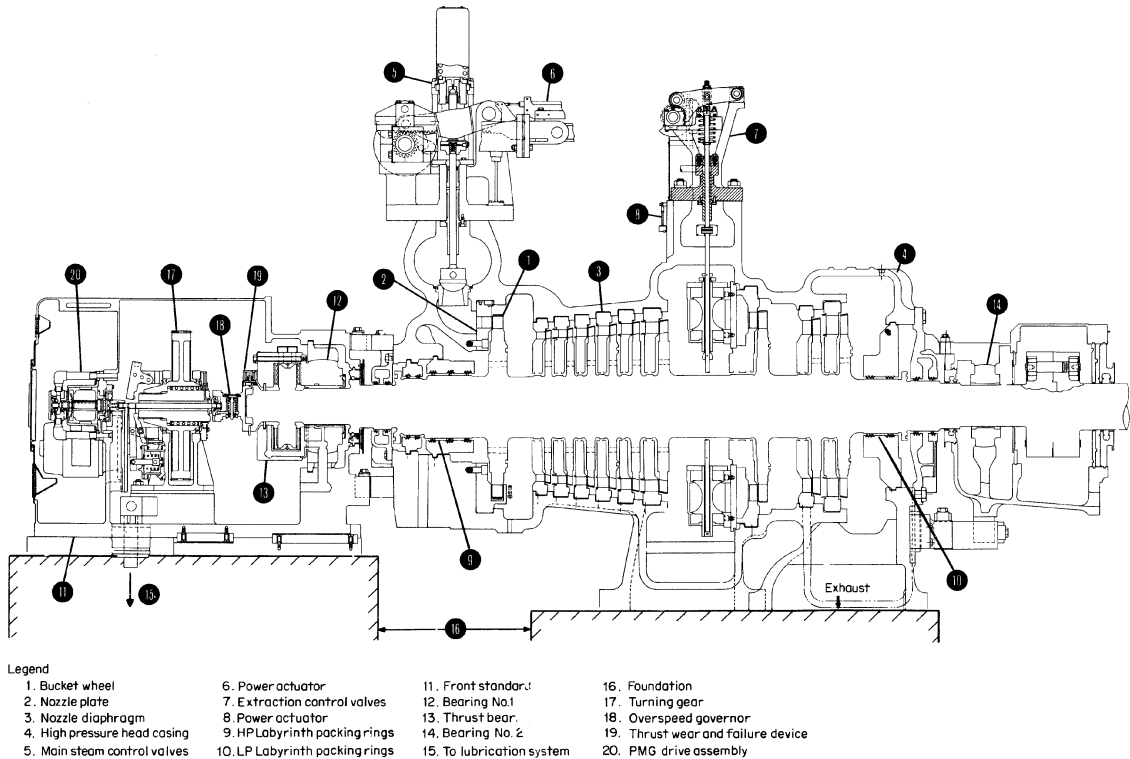


FIGURE 6-5 Cross section of a modern single-automatic-extraction noncondensing steam turbines showing construction details. (*General Electric.*)

and stationary blading (buckets and nozzles); (4) casing to contain the stationary parts and act as a steam pressure vessel; (5 to 8) controlling and protective valves, piping, and associated components to accept and control the steam admitted to the steam path; (9 and 10) packing and sealing arrangement to prevent steam from escaping into the surrounding area; (11) front standard which houses lubrication, control, and protective equipment and supports part of the casing; (12 to 14) set of journal and thrust bearings to support the rotating elements and absorb all static and dynamic rotor loads; (15) lubrication and hydraulic system for supplying bearing lubrication and (when applicable) generator seals, control, and protective oil requirements; (16) supporting foundation on which the major stationary parts rest; and (17 to 20) various accessory components, such as turning gear, control and protective components, drain valves, etc., as required by the specific application.

Turbines are constructed chiefly of carbon, alloy, and stainless steels. The rotor may be a single forging, fabricated from a shaft and separate wheels, or constructed of forged elements welded together. The buckets forming the rotating portion of the steam path are generally machined from solid stock and attached by pins, or grooves called “dovetails,” to the wheels. The stationary steam path is built up of diaphragms with nozzles mounted in the heavy, two-piece casing (usually cast steel), which is bolted together on a horizontal joint. If the unit is condensing or has a low back pressure, the exhaust casing may be made up as a separate assembly and bolted to the main casing through a vertical joint. To minimize thermal stresses in high-temperature applications (950 to 1050°F), a double shell casing may be used. The rotor may sometimes be made in two pieces and coupled together, particularly in the case of the larger condensing double-flow units. A solid or flexible coupling may be used to connect the turbine rotor to its load.

Labyrinth-type packing rings, consisting of high and low teeth, are arranged at the ends of the steam path to inhibit steam from escaping into the surrounding area. (Similar packing is used at each diaphragm, and particularly at stages having control valves, to prevent excessive leakage from one stage to another within the steam path.) Associated with the external packing is a seal system which draws a vacuum to exhaust a mixture of leaking steam and air and thus prevents any steam from leaking into the surrounding room.

Bearings for supporting the turbine rotor are located in pedestals at either end and consist of journals and a thrust assembly. Normally, when steam flows through the turbine, thrust is developed in the direction of steam flow. However, unusual operating conditions or configurations often cause a thrust reversal. Therefore, it is usually necessary to provide an “active” thrust bearing for normal loading and an “inactive” thrust bearing for reverse loading.

The control-valve gear-activating equipment in a turbine usually is mounted on top of the turbine casing at the stage where steam is to be admitted. There are at least as many valve-gear assemblies as there are control stages, sometimes more if a lower valve-gear assembly is required for passing the flow. Protective or emergency valves are generally located off the machine, near the associated steam piping. Control, protective, and accessory components are often located in part in the front standard, at the pedestal housing the first journal and thrust bearing assemblies. The unit is usually supported on its foundation at the front standard and at the exhaust casing. The coupled generator shares similar foundation supports.

6.1.7 Turbine Control and Protective Systems

Steam turbines require a number of systems and components to provide control and protective capability. These may be divided into two functional categories: (1) primary control systems and (2) secondary and/or protective control systems.

Primary Control Systems. *Primary control systems* may be further subdivided into the following elements: control valves and associated operating gear, speed/load control, and pressure control. *Secondary or protective systems* consist of overspeed limiting devices, emergency valves, trip devices, and associated alarm devices.

Control-Valve Gear. Most modern turbine-generators use steam-admission control-valve designs which are as efficient as practical, in terms of pressure drop and throttling losses. Most popular are the ball-venturi valves used in the inlet stages of modern high-pressure units. The use of multiple valves, with the efficient venturi seat configuration and the tight-seating ball valves, permits partial-arc nozzle admission to the turbine with good part-load efficiency and a sequential opening action which produces nearly linear flow curves. These valves may be opened by one of two basic means: bar lift, with valves sequenced by stem lengths, or cam-operated by levers and rollers, to linearize the inherently nonlinear flow characteristics of ball-venturi valves. A common variation of the ball-venturi valve gear once widely used is the poppet-valve gear, with beveled valves and seats. This arrangement is not as efficient as a ball-venturi valve gear and is not presently very popular.

Another commonly employed valve gear, particularly on lower-pressure high-volume-flow applications, is the double-seated spool valve gear. This valve is not very efficient but passes high-volume flow and can be programmed to open in a manner similar to a ball-venturi valve.

A third scheme used somewhat in the past, but now less popular, is the grid valve, which consists of two plates with specially shaped holes arranged so that when one is rotated relative to the other, the flow area is developed as the holes coincide. High volume flow and short physical span are the grid valve's strong points, but its efficiency and accuracy are not good, and operating forces are high.

Speed/Load Control Systems. After a choice has been made from the types of valves available for steam admission to the turbine, a means must be provided for positioning these valves to obtain basic speed/load control. The primary requirement is the maintenance of an accurate, predetermined rotating speed, since all turbines are designed to operate at a specific speed or over a specific range of speeds. Every turbine, therefore, has some type of speed governor or, more generally, “speed/load control system.” Its purpose is to maintain a relationship between actual turbine speed and some reference value, over a wide range of load torques.

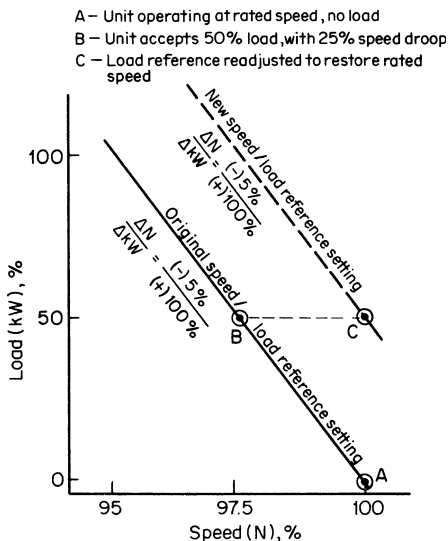


FIGURE 6-6 Steady-state speed/load regulation for a given reference setting in the speed/load system of a steam turbines. (*General Electric.*)

droops proportionally with increasing load, on a steady-state basis. The system should be so designed that a steady-state error in speed is required to provide the command signal to move the control valve gear to accept the required load.

If such a unit operates independently, the speed/load characteristic will be as shown by the solid line in Fig. 6-6 (point A). If the unit is tied to a system much larger than itself, and the same system load change occurs, obviously the effect on the unit will be much less, and speed will not vary as much. The system is said to be “stiff” compared to the unit. Since speed accuracy is very important if the operating unit is isolated, any speed droop experienced with a load change (point B) must be corrected by changing the speed/load reference setting. This is illustrated by the dashed line of Fig. 6-6, where a 50% load change was followed by a reference correction to restore rated speed (point C). If an operating unit is tied to a “stiff” system, and it must accept more of the system load, a similar adjustment will cause it to pick up load with no change in speed, as the dashed line shows.

Manual speed/load reset, therefore, permits a unit, whether isolated or tied to a system, to be set to hold speed, or carry load, as the operator desires. However, if such a unit is to operate for long periods of time, and under varying load conditions, manual load reset is an inadequate solution to the problem of maintaining speed accuracy. In such cases, the use of an automatic reset device or “speed corrector” to provide isochronous control is common.

Figure 6-7 shows a very simple form of speed/load control system: a mechanical speed governor suitable for very small turbines. This type of governor uses a spring-load mechanical flyball mechanism connected to a throttling valve to directly control steam admission to the turbine. On larger units, where the forces required are too high for direct operation, a hydraulic relay governing system, as shown in Fig. 6-8, is used. In this arrangement a centrifugal flyball-type governor is connected through linkage to a double-spoiled pilot valve. Oil is admitted to the pilot valve, so that when the valve moves, it ports fluid either into or out of an operating cylinder as required. The motion of the cylinder restores the pilot valve, through another linkage, to maintain a stable relationship between the pilot valve and its cylinder. Available force for operating the control valves is multiplied many times with this arrangement. On units larger than about 1000 kW, a mechanical hydraulic control system having two or more such hydraulic relays or amplifiers is used to multiply available force and to operate multiple control-valve gear systems.

Land turbines used for power generation generally operate at a specific rated speed whereas marine and mechanical-drive turbines, because of the inherent coupling characteristics between rotating blades and fluids, operate over a range of speeds.

In most of the Western Hemisphere, the accepted operating frequency for turbine-generator machinery is 60 Hz. Such units having a 2-pole generator must, therefore, operate at 60 r/s, or 3600 r/min. A 4-pole unit operating at 60 Hz will rotate at half the speed of a 2-pole unit, or 1800 r/min. Most units in the United States operate at either 1800 or 3600 r/min. In most of the remainder of the world, the accepted electric frequency is 50 Hz, with common operating speeds of 1500 or 3000 r/min. In order to understand steam-turbine speed/load control, it is helpful to consider the example of constant-speed/load-based utility or industrial units.

Figure 6-6 shows the relationship designed into the speed/load control system of most such units built in the United States. The commonly accepted speed “droop,” with load, for such a system is (–) 5% for 100% load change, based on a given speed/load reference setting. Note that speed

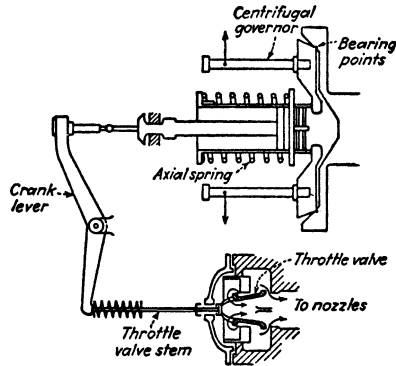


FIGURE 6-7 Mechanical governor for small turbines.

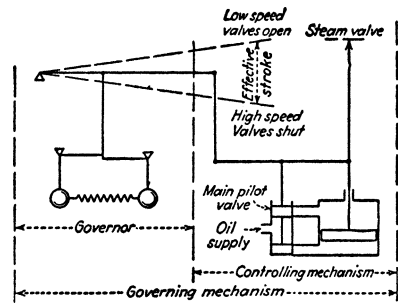


FIGURE 6-8 Governor with hydraulic power amplifier.

The electrohydraulic control system offers greater accuracy, higher operating forces, remote and centralized control capability, and more options and flexibility than any previous system. The speed/load control system in Fig. 6-9, consists of (1) a permanent magnet generator or digital-type reluctance pickup to provide a shaft-speed signal, (2) electronic circuitry for comparing the speed signal with a reference signal, (3) a high-gain servo valve to convert the resulting electric signal to a hydraulic signal, (4) a valve-gear power-actuator assembly capable of operating on high-pressure hydraulics on receipt of the servo-valve signal, (5) a feedback transducer on the power actuator to restore the servo valve to a stable condition when the desired valve position is reached, and (6) a high-pressure hydraulic system to provide the force required.

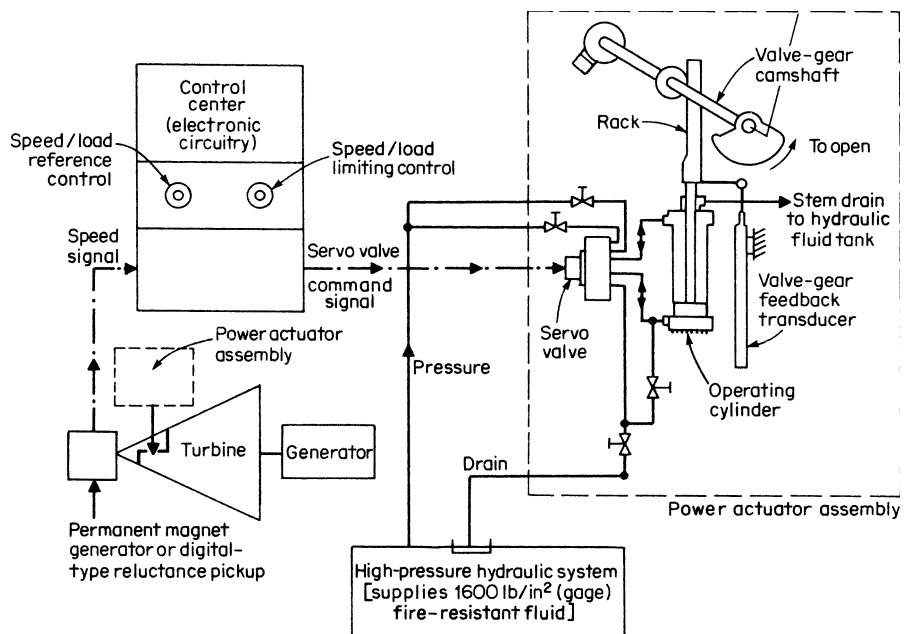


FIGURE 6-9 Schematic diagram of a basic electro-hydraulic speed/load control system. (General Electric.)

Other types of speed/load control systems have been applied to turbines from time to time. These include pneumatic, hydraulic, or electric devices. However, the two most common systems for turbine control are the mechanical hydraulic control (MHC) and electrohydraulic control (EHC) systems described. Another version of the EHC system was developed in the late 1960s to provide bridge control on marine turbine applications.

Pressure Control Systems. A second major area of control technology on steam turbines deals with process control. In industrial power plants particularly, it is often economical to generate and control several process flows, using steam from available steam turbines. As in the case of speed/load control, when a process is to be controlled, a definite relationship or “regulation” is established between the flow to be supplied by the turbine and the pressure. However, the possible options in process-pressure-control management are much greater than the speed/load control options described. The most common application control is for an extraction or exhaust flow from a turbine, which is to be controlled accurately in pressure and used in an industrial process. For this purpose, automatic-extraction and exhaust pressure control systems have been designed, using both the MHC and EHC technologies. Occasionally, particularly on waste-heat boiler applications, there is a need for initial pressure control as well.

Figure 6-10 is a greatly simplified schematic representation of a mechanical hydraulic control system on a single-automatic-extraction condensing turbine. The unit consists of two turbines, an HP and an LP section (each supplied by a separate valve gear), on one shaft. A flyball speed governor is used to move the two sets of valves to control speed, or load, and a bellows-type pressure governor is employed to sense process pressures and move the valves in opposite directions to control process flow and pressure. (Actual hardware required for these actions would, of course, include either mechanical hydraulic relays and linkage or electrohydraulic components.) The system is usually so designed that load and process flow variations can be satisfied at the same time with a minimum of interaction between the two variables.

Often the need exists as well for control of exhaust pressure on a noncondensing turbine-generator. In this case, since the number of variables which can be controlled is only equal to the number of control-valve stages, one variable must be sacrificed. Usually, the unit is tied to a “stiff” electrical

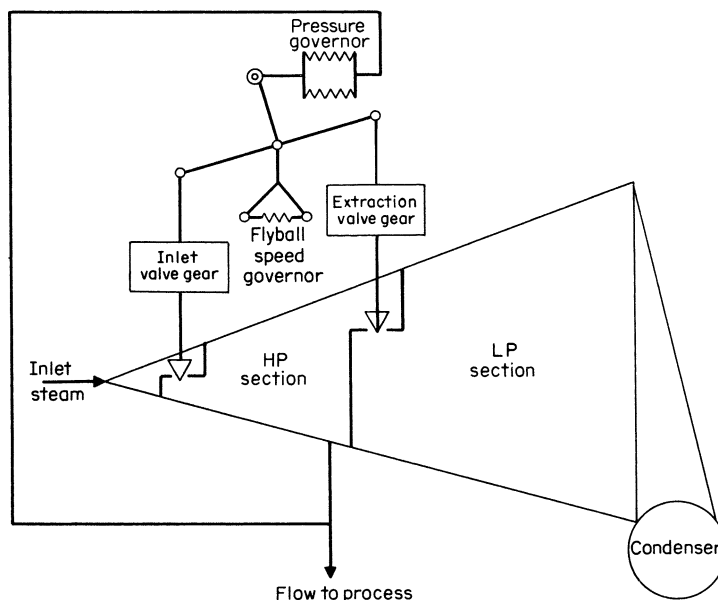


FIGURE 6-10 Simplified schematic diagram of a speed and pressure control system for a single-automatic-extraction condensing turbine. (General Electric.)

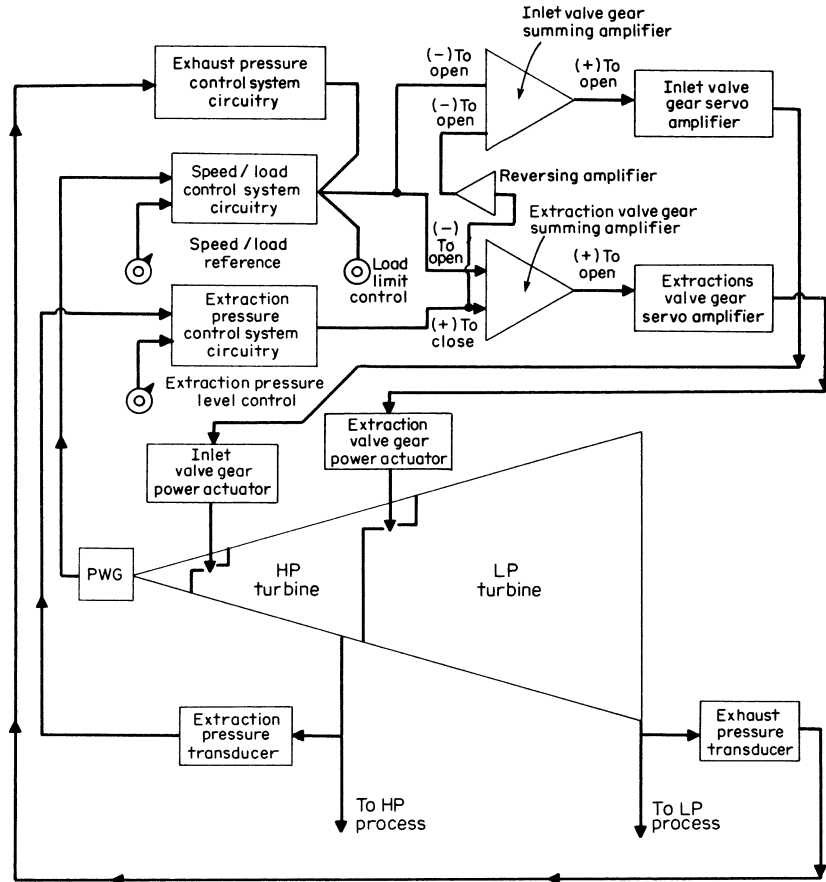


FIGURE 6-11 Schematic diagram of a modern electro-hydraulic control system for a single-automatic extraction noncondensing turbine. (General Electric.)

system, and speed/load control is sacrificed in favor of exhaust pressure control. Figure 6-11 shows a modern electrohydraulic control system for a single-automatic-extraction noncondensing turbine capable of controlling two process pressures for industrial needs.

6.1.8 Lubrication and Hydraulic Systems

Forced-feed lubrication of turbines and generator bearings is normally used on units above approximately 200 hp in size. On such units, the lubrication system is sometimes used to supply low-pressure seal oil for a hydrogen-cooled generator as well. Also, it often is used to supply the higher-pressure oil for the turbine control and protective systems. This is normally the case on units having a mechanical hydraulic control system and operating on turbine oil at a pressure of 250 lb/in² (gage) or less. On units having electrohydraulic control systems, operating at higher hydraulic pressures up to 3000 lb/in² (gage), and using fire-resistant fluids, a separate hydraulic-fluid power unit supplies all fluid for the control systems and usually for the protective systems as well.

6.1.9 Oil-Seal and Gas-Cooling Systems for Hydrogen-Cooled Generators

For steam turbine-generators rated up to about 40,000 kW, the electrical windings are generally cooled by air. However, above this size range, most units have hydrogen-cooled generators. Liquid cooling with hollow conductors is used on the largest units, above about 300,000 kW. Hydrogen cooling is employed because hydrogen has a thermal conductivity nearly 7 times that of air, and a density only one-fourteenth that of air. This permits reduction of windage losses and increased cooling, thereby increasing load-carrying capability for a given size of hardware.

A shaft sealing system is required to properly seal the hydrogen for cooling larger units. Oil is the sealing medium and is pressurized above hydrogen gas pressure so that it leaks across the seals to a cavity which is a receiving area for the hydrogen leaking out of the generator casing. The hydrogen-oil mixture is then scavenged through a dryer to a hydrogen control cabinet which monitors the pressure, temperature, and purity of the gas mixture, in order to maintain a safe hydrogen concentration in the generator.

6.1.10 Miscellaneous Steam-Turbine Components

In addition to the systems and components discussed in the earlier sections, steam turbines often have a number of accessory components which are important to their operation. A turning gear is provided on units rated larger than approximately 10 MW, to slowly rotate the turbine shaft before the unit is started and after it is shut down. This action helps prevent rotor bowing due to unequal heating or cooling of the rotor.

Another device of some importance is a lifting gear, for assembly and disassembly of the unit during installation and outages.

A set of turbine supervisory instruments is often included with a steam-turbine package. Typically monitored items are shaft vibration, differential thermal expansion between the casing and the rotor, expansion of the casing, eccentricity while on turning gear, thrust bearing position, speed of rotation, acceleration, control-valve position, and various other items as required by design and/or customer needs.

6.2 STEAM-TURBINE APPLICATIONS

6.2.1 Central-Station Turbines

A 60-MW, 3600-r/min *nonreheat steam turbine* is typical of those installed in smaller utility plants. Steam flows into the steam chest and through the control valves to the first-stage nozzle. After expanding through a Curtis-type, 2-row control stage, the steam flows through 16 more Rateau (impulse) stages to the exhaust. During the expansion, some steam is bled off at four or five extraction points for feedwater heating. Larger-rated units, such as those used in combined-cycle plants, require double flowing of the last five or six stages in order to provide the last-stage annulus area necessary to maintain a low leaving loss.

A 600- to 800-MW *tandem-composed single-reheat steam turbine* is typical of the type used in large fossil-fired central stations. Steam conditions are predominantly 2400 lb/in² (gage), 1000/1000°F, but some applications are at 3500 lb/in² (gage) and a few utilize double reheat as well. Steam enters the high-pressure turbine element through four pipes leading from the off chest-control valves to the nozzle box and the double-flow first stage. The flow is expanded through six more impulse stages before exiting from the HP casing to the reheater. The reheated steam enters at the center of the intermediate turbine and expands through seven double-flow intermediate stages before exhausting to the crossover pipe. The crossover feeds the steam to the four-flow LP elements where it is expanded to completion through six more stages before exhausting to the condenser.

A typical *nuclear steam turbine* has a capacity of 1000 to 1300 MW. Steam enters the double-flow high-pressure element at the left at 1000 lb/in² (gage), 546°F, and exhausts at about 200 lb/in² (abs) to the two large combined moisture-separator reheaters which straddle the three double-flow low-pressure elements. After the moisture is removed and the steam slightly reheated, it is passed to the six-flow

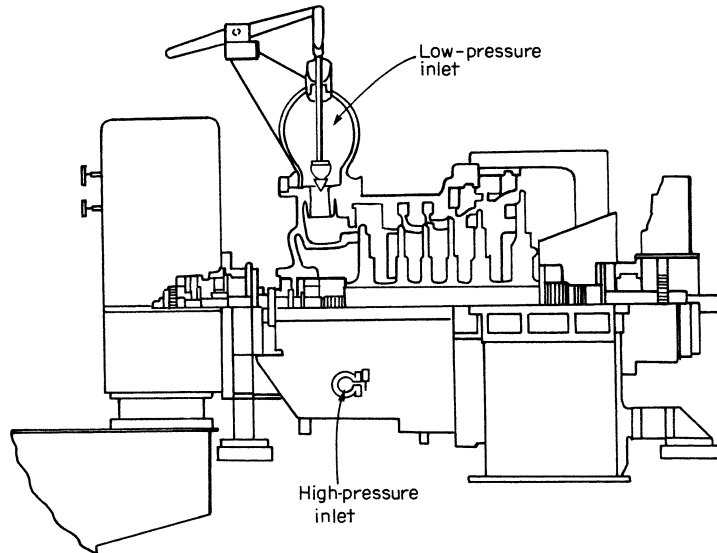


FIGURE 6-12 A 10,000 hp, 5500-r/min steam generator-steam feed-pump-drive turbine for nuclear application. (General Electric.)

low-pressure element where it is expanded to completion. The low superheat available with the light water reactors and the large ratings encountered require the use of 1800-r/min machinery to keep blade speeds low, reducing the erosion from moisture, and to provide the large flow areas which are more easily obtained using larger wheel diameters. The moisture-separator reheater is provided to decrease the erosion in the low-pressure elements and improve their performance. The *condensing boiler-feedwater-pump-drive turbine* was introduced in the 1960s, and became accepted rapidly because the increase in condensing annulus area served to improve both the heat rate and capacity of the station. Figure 6-12 illustrates a 10,000-hp straight condensing boiler-feed-pump-drive turbine for a nuclear application. Steam is extracted from the main unit cycle after it has gone through the moisture separator-reheater and is delivered to the inlet at approximately 150 lb/in² (abs) and from 0 to 100°F superheat. It is expanded to completion in the six stages. For operation at light load, steam is taken from the main steam header and sent to the high-pressure inlet in the lower half of the first stage.

6.2.2 Industrial Steam Turbines

In many industrial plants, particularly those in the pulp-and-paper, petrochemical, and related industries, the need exists for large amounts of electric power and process steam at various pressure levels. In this type of situation, industrial users can justify generating their own power and charging a large part of the cost to the process, because the steam is also needed. Plants have historically been built and expanded with various condensing and noncondensing extraction turbine types, as the process requires, and in sizes ranging from 1 to 200 MW.

6.2.3 Variable-Speed Turbines

The steam turbine is used extensively as the prime mover for ship propulsion at ratings above 10,000 shp. The cross-compound design is almost universal as it provides emergency capability for getting back to port as well as providing two pinions which divide the load on the low-speed gear, reducing gear weight. The major applications are in high-powered, high-utilization ships such as tankers and container ships. They are used almost exclusively in naval combat ships (aircraft carriers and nuclear submarines) as well as for large auxiliary supply ships.

Applications are predominantly nonreheat, but because of the steadily rising cost of fuel, reheat applications are gaining popularity.

Industry uses *mechanical-drive turbines* for a wide variety of purposes. These turbines drive paper machines; blast furnace blowers; and ethylene, ammonia, and liquefied natural-gas plant compressors because of their ability to follow the speed-output characteristics of this type of equipment without loss of efficiency from throttling, recirculation, or use of fluid couplings.

6.2.4 Special-Purpose Turbines

Turbines have been designed and built for many unusual applications. They have been built for use with working fluids other than steam in refineries and petrochemical plants. Mercury has been used as a working fluid in the *binary steam power plant*. Steam turbines have been tried in steam-locomotive applications.

Steam turbines utilizing *geothermal steam* have operated for many years in Italy, and more recently in New Zealand and the United States. The Geysers fields in California contain a high grade of geothermal energy as steam, available at the wellhead at about 100 lb/in² (gage). The high cost of liquid fuels has increased the attractiveness of the much more extensive geothermal brine sites as well. The geothermal-steam turbine is required to pass a much greater volume flow of steam per kilowatt generated than the central station plant. The presence of impurities in geothermal steam requires much more extensive use of alloys in the steam path and protection against moisture in the steam. The development of brine fields, which contain 300 to 500°F liquid in the wells, will require development of turbines with much larger volume-flow capacities to recover the low-level energy available. Alternative development may utilize heat exchangers which will transfer the energy to other fluids, such as isobutane, for expansion in the turbine.

6.3 STEAM-TURBINE PERFORMANCE

6.3.1 Rankine-Cycle Efficiency

The steam turbine constitutes the expansion portion of a vapor cycle, which requires separate devices, including a boiler, turbine, condenser, and feedwater pump, to complete the cycle. This vapor cycle for steam power plants is commonly called the Rankine cycle (Figs. 6-13, 6-14) and is less efficient than the Carnot cycle because the exhaust vapor is completely liquefied to facilitate pumping, and because superheat is added at increasing temperature. The work of the cycle is equal

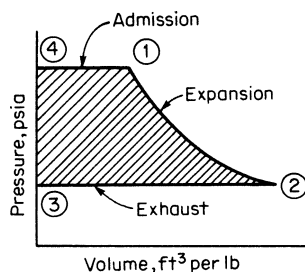


FIGURE 6-13 Pressure-volume diagram for the Rankine cycle; Phase a4-1, constant-pressure admission; phase 11-2, complete isentropic expansion; phase 2-3, constant-pressure exhaust. Crosshatched area represents the work of the cycle.

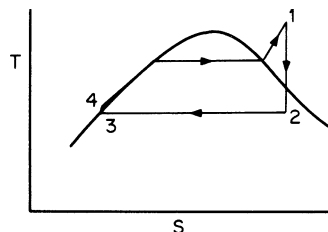


FIGURE 6-14 T-S diagram; nonreheat, nonextracting turbine cycle.

to $h_1 - h_2$ minus the small pump work $h_4 - h_3 = v_3(P_4 - P_3)/J$ required, and the heat added to the cycle is equal to $h_1 - h_4$. Therefore

$$\text{Rankine-cycle efficiency} = \frac{(h_1 - h_2) - v_3(P_4 - P_3)/J}{h_1 - h_4} \quad (6-4)$$

Cycle efficiency is not commonly used when comparing plant efficiencies because it is only indirectly determined, compared with heat rate, which can be quickly measured. In central-station practice, the station heat rate defines the heat required from fossil fuel, reactor energy, waste gases, etc., per kWh of station output (gross electric output minus all auxiliary power required within the plant). For example, a plant with a station heat rate of 10,000 Btu-fuel/kWh has a seal cycle efficiency of

$$\text{efficiency} = \frac{\text{output}}{\text{input}} = \frac{3412.1}{10,000} \times 100 = 34.1\% \quad (6-5)$$

Power plants for marine propulsion drive commonly use the “ships all-purpose fuel rate” (lb fuel/shp-h) as a measure of the plant’s efficiency. Besides accounting for all losses required to generate the propeller-shaft output work, this fuel rate includes the requirements for the ship’s “hotel” electric load, freshwater evaporators, and steam for heating and unloading cargo and cleaning tanks. A typical ship’s fuel rate of 0.45 lb fuel/shp-h based on 18,500 Btu/lb fuel would indicate cycle efficiency for the ship as

$$\text{Cycle efficiency} = \frac{2544.1}{0.45 \times 18,500} \times 100 = 30.6\%$$

In the process industries, such as paper and petrochemical, large amounts of steam are used. Considerable by-product power can be generated by raising the boiler pressure above the process pressure and expanding the steam through a noncondensing turbine before exhausting it to the process. In this cycle, no heat is rejected because the exhaust steam is required for process and the thermodynamic cycle efficiency of this power is affected only by the boiler efficiency, the auxiliary losses chargeable to the power generation (mostly extra boiler-feed-pump work), and the mechanical and electrical losses of the turbine and generator.

The station heat rate of such by-product power generation ranges from 3900 to 4500 Btu/kWh, depending on the size of plant and the boiler efficiency, and cycle efficiencies of 80% to 85% are normal. This heat rate varies very little with turbine efficiency because energy not used to generate power is used for process. However, it is necessary to define the kilowatts generated per unit of heat to process in order to evaluate the influence of turbine efficiency or the initial steam conditions selected. Guaranteed steam rates are normally provided to evaluate the efficiency because they can be directly compared with the theoretical steam rate (TSR).

6.3.2 Engine Efficiency

The station heat rate is used to measure power plant performance, but it is of little use in evaluating the specific pieces of equipment in the cycle. The engine efficiency of the steam turbine defines its actual performance to the ideal performance. The Rankine-cycle work of the turbine is most conveniently obtained by use of the Mollier diagram (Fig. 6-15), where

$$\Delta W = h_1 - h_2 \quad (6-6)$$

and ΔW = Rankine-cycle work in Btu/lb, h_1 = steam enthalpy at throttle in Btu/lb, h_2 = steam enthalpy at exhaust in Btu/lb,

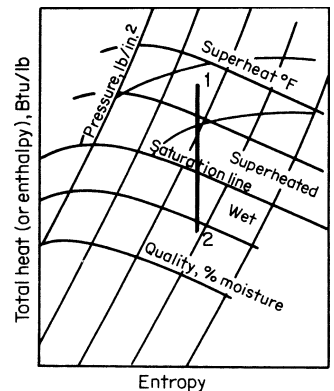


FIGURE 6-15 Steam chart (Mollier diagram.)

and h_1 and h_2 are at the same entropy (vertical line). The actual work of a real turbine is less than the ideal Rankine-cycle work, with engine efficiency defined as

$$\text{Engine efficiency} = \frac{\text{actual work, Btu/lb}}{\text{Rankine-cycle work, Btu/lb}} \quad (6-7)$$

6.3.3 Theoretical Steam Rates

The theoretical steam rate in lb/kWh for the Rankine-cycle work is expressed as

$$\text{TSR} = \frac{3412.14}{h_1 - h_2} \quad (6-8)$$

and the actual steam rate as

$$\text{ASR} = \frac{\text{TSR}}{\text{engine efficiency}} \quad (6-9)$$

The actual output of the turbine will be less than the isentropic work because of losses from nozzle and bucket friction, packing leakage, windage, carryover and exhaust losses, bearings, radiation, and throttling. The efficiency of the turbine, including its losses, is sometimes represented by the term η_{evpm} , to help identify the losses which are included in the efficiency. The subscript e denotes exhaust loss (significant on condensing units only), v denotes valve loss (control valves only), p denotes external packing loss (valve stems and shaft end packings), and m the mechanical losses. The efficiency of the generator or gear is represented by η_g .

This or similar terminology enables turbine designers to identify a particular efficiency for discussion. The stateline efficiency η represents the actual Mollier-chart expansion line of a particular turbine for given inlet steam conditions and exhaust pressure without any exhaust loss included.

The η_{evp} efficiency represents the internal efficiency corrected to include the exhaust loss, a "mean" of the control-valves pressure-drop loss, and the packing loss for the point in question. When an electric generator is the driven equipment, the overall engine efficiency η_{evpmg} is used to determine the throttle flow necessary to produce a given electric output (or vice versa).

6.3.4 Condensing-Turbine Efficiencies

Table 6-2 defines some approximate overall efficiencies of typical small straight condensing turbine-generators. The application of small turbines without regenerative feedwater heating is rare in central-station practice but is still found in waste-heat applications, process plants where feedwater heating is supplied by other sources, and increasingly in combined cycles where the stack gas is used to heat feedwater.

For turbines used in central stations, there are more satisfactory methods (see Bibliography at the end of this section) for predicting turbine efficiency, which take into account the many variables of the steam path and exhaust size and allow for inclusion of the regenerative cycle and reheat cycle in heat-balance calculations.

6.3.5 Regenerative Cycle

Steam can be extracted at several stages in the turbine to heat feedwater being returned to the boiler. In the Rankine cycle (Fig. 6-14), it was shown that the feedwater was heated from h_4 to h_1 in the boiler. By raising the temperature h_4 entering the boiler close to the saturation temperature in the drum, less fuel will be consumed in evaporating each pound of steam to h_1 conditions. The heat in the extracted steam is added to the feedwater without loss, and the heat rejected to the condenser decreases as extraction flow increases. The kilowatts do not decrease inversely with extraction flow, however, as partial expansion is made down to the extraction stages. The result is an improvement in heat rate.

TABLE 6-2 Typical Efficiencies of Straight Condensing Turbine-Generators

kW Rating	Base efficiency, f_1				
	5000	10,000	15,000	20,000	30,000
250	0.743	0.766			
400	0.733	0.757	0.769	0.777	
600	0.720	0.748	0.763	0.772	0.776
850		0.742	0.758	0.768	0.773
1250			0.754	0.765	0.770
Correction for initial superheat					
°FS	0	100	200	300	400
f_2	0.95	0.98	1.00	1.017	1.030
Correction for exhaust pressure					
P_f , inHg (abs)	1.0	1.5	2.0	3.0	
f_3	0.98	1.00	1.01	1.02	

Example: 15,000-kW turbine-generator

Steam conditions: 850 lb/in² (gage), 900°F_{TT}, 2.5 inHg (abs)

TSR = 6.42 lb/kWh Superheat = 372.8°F

100% load: $\eta_{evpmg} = f_1 \times f_2 \times f_3 = 0.758 \times 1.026 \times 1.015 = 0.789$

100% ASR = $\frac{6.42}{0.789} = 8.14$ lb/kWh

Throttle flow $F_r = 15,000 \times 8.14 = 122,100$ lb/h

Source: Medium Steam Turbine Department, General Electric Co.

The actual heat rate of a regenerative cycle must be determined from a heat balance prepared by using the extraction conditions available from the turbine and the heater characteristics as specified. Table 6-3 shows the influence of heater type, temperature difference, and piping pressure drop on the gross heat rate of a 50,000-kW unit.

6.3.6 Reheat Cycle

Practically all large (over 100,000 kW) central-station plants built since 1950 have been of the reheat type. In this type of fossil-fuel plant, the steam is expanded in the turbine down to about 25% of the initial pressure, then it is sent through a reheater where it is resuperheated back up to the original initial temperature (usually 1000°F) and returned to the turbine where it is expanded to completion. In general, reheating improves the heat rate by about 5% of which only 2% is due to a higher average cycle temperature and about 3% comes from improved turbine internal efficiency due to reduced moisture and increased reheat factor.

6.3.7 Gross and Net Heat Rates

The heat-balance cycle for the turbine, condenser, feedwater pump, and heaters usually defines the gross heat rate of the cycle when a motor-driven feed pump is used. The pump work on the feedwater is included, but the power consumed by the pump is not. When a turbine-driven boiler feed pump is used in the cycle (frequently in units above 300-MW rating), the steam for the feed-pump turbine is expanded in the main unit down to the crossover to the low-pressure elements (about 100 to 200 lb/in² [abs]), where it is sent to the pump turbine. In these cases, the pump power required is included in the heat balance, and the heat rate calculated is called the net heat rate. In this case, only the losses from the boiler and auxiliaries (excluding the feedwater pump) must be accounted for to obtain the station heat rate.

TABLE 6-3 Heat-Rate Variation with Heater Cycle of 50,000-kW Nonreheat Turbine [1250 lb/in² (gage), 950°F_{TT}, 1.5 inHg (abs)]

Base cycle HR = 8852 Btu/kWh, FWT = 430°F

	Heater no.					Δ Heat rate, Btu/kWh
	5	4	3	2	1	
TTD, °F	5	5	Open	5	5	Base
DCTD, °F	10	10	Htr.	10	10	Base
% ΔP	5	5	5	5	5	Base
Variation in terminal temperature difference, TTD						
	0	5		5	5	-10
	-5	5		5	5	-20
	5	5		5	10	+6
	10	10		10	10	+27
Variation in drain cooler temperature difference, DCTD						
	5	5		5	5	-2
	20	20		20	20	+5
	10	10		10	20	+2
Variation in pressure drop to heater, % ΔP						
	0	0	0	0	0	-24
	10	10	10	10	10	+26
	10	5	5	5	5	+10
Variation from drain cooled to pumped or cascaded drips						
	10	10		10	C	+12
	C	C		C	C	+30
	10	PD		10	PD	0
	10	C*		10	10	+32

*Cascaded to no. 2 heater.

Nomenclature: TTD = heater terminal temperature difference; DCTD = drain cooler temperature difference; % ΔP = pressure drop from turbine flange to heater shell; C = cascaded heater drains; PD = drains pumped forward.

Note: $t_{c} = (t_{F} - 32)/1.8$.

Source: Medium Steam Turbine Department, General Electric Co.

Table 6-4 shows typical values of gross and net heat rates at rated load for nonreheat and reheat units of typical ratings and inlet steam conditions. Exhaust pressure is 2.0 inHg (abs) in all cases, motor-driven feed-pump drive efficiency is 90%, pump efficiency is 78%, and the exhaust annulus area is normal for the rating.

The overall station heat rates of these applications are 15% to 25% greater than the net heat rate, depending on the steam-generator (boiler) efficiency and other auxiliary losses.

6.3.8 Nuclear Cycles

The turbines used in light-water nuclear cycles do not have the same freedom of steam conditions as fossil-fired cycles. The nuclear steam supply limits initial steam pressure to approximately 950 lb/in² (gage) at a saturated steam temperature of 540°F. The initial costs of these plants is so great that only the largest can be economically justified, and ratings are limited to about 800 MW minimum. Crossover pressures range from about 150 to 200 lb/in² (abs), and regenerative feedwater heating improvement is optimized by using about six heaters. These constraints and optimizations of the nuclear power-plant cycles have resulted in a rather narrow band of heat-rate fluctuations. At rated

TABLE 6-4 Typical Heat Rates of Nonreheat and Reheat Turbine-Generators

	Steam conditions			Last-stage buckets			Performance at 2.0 inHg (abs)					
	P_{0^*} lb/in ² (gage)	T_{0^*} °F	T_{RH^*} °F	No. rows	Annulus area, ft ²	Boiler feed-pump drive	Number of feedwater heaters	FFWT, °F	Gross heat rate, Btu/kWh	Net heat rate, Btu/k Wh	Throttle SR, lb/kWh	Condenser SR, lb/kWh
15	600	825		1	12	Motor	3	390	10610	10700	10.05	7.52
25	850	900		1	14	Motor	4	365	9730	9850	8.73	6.69
40	1250	950		1	19	Motor	5	444	9240	9410	8.86	6.23
50	1250	950		1	26	Motor	5	429	9080	9240	8.56	6.10
75	1250	950		1	41	Motor	5	437	8890	9040	8.44	5.96
75	1450	1000	1000	1	33	Motor	5	431	8320	8450	6.69	4.92
75	1800	1000	1000	1	33	Motor	5	448	8140	8290	6.70	4.80
100	1800	1000	1000	1	41	Motor	5	458	8110	8270	6.77	4.80
150	1800	1000	1000	2	66	Motor	6	448	7970	8130	6.51	4.66
200	2400	1000	1000	2	82	Motor	7	461	7750	7950	6.39	4.49
350	2400	1000	1000	2	132	Motor	7	473	7750	7950	6.49	4.45
350	2400	1000	1000	2	132	Turbine	7	473		7890	6.62	4.27
500	2400	1000	1000	4	222	Turbine	7	473		7830	6.57	4.26
700	2400	1000	1000	4	264	Turbine	7	473		7870	6.58	4.27
1000	3500	1000	1000	6	334	Turbine	7	504		7730	6.71	4.04

Note: 1 ft² = 0.0929 m²; 1 lb/m² = 6.895 kPa; $t_c = (t_{rf} - 32)/1.8$; 1 in = 25.4 mm.

Source: Medium Steam Turbine Department, General Electric Co.

load and 2.0 inHg exhaust pressure, the turbine net heat rate varies from about 9800 Btu/kWh at an exhaust loading of 1500 kW per square feet of annulus area up to about 10,000 Btu/kWh at 2000 kW/ft² of exhaust loading.

Turbines used in the high-temperature, gas-cooled reactor cycle operate at steam conditions, including reheat similar to those used in fossil-fired plants. Because of the low moisture content and lower volume flow required, 3600-r/min turbine design speed can be utilized as well. At 2400 lb/in² (gage), 1000/1000°F, 2.0 inHg (abs) design conditions, a heat rate of about 8300 Btu/kWh can be obtained, including the power required for gas recirculation. This represents a 15% to 17% improvement in cycle efficiency when the first plant becomes operational.

6.3.9 Combined Cycles

Improvement in the steam cycle has been rapid. The advancement of steam conditions, regenerative feed heating, reheating, and size of unit has brought the overall station heat rate down from 16,000 Btu/kWh to less than 8800 Btu/kWh on the best station.

The gas turbine developed very rapidly as a prime mover because of the improved steam cycle. During the 1950s, several exhaust-fired combined-cycle power plants were built, utilizing the exhaust gas from the gas turbine as the air supply for a fired main steam generator. After the Northeast Blackout of 1965, a large number of gas turbines were installed in the United States to serve as black start and peaking capacity units. As a result, they have become well established and accepted as a prime mover for peaking capacity.

Combined-cycle interest was renewed with the development of non-radiant-heat recovery steam generators, and the electric generation in combined-cycle plants changed from 80% to 90% steam-cycle power to 70% gas-cycle power. Since 1970, utilities have installed increasing numbers of this breed of combined-cycle plant, as they offer low initial cost, consume about one-third of the water used by straight steam plants, and provide a station heat rate 5% to 10% better than the most efficient steam plants. The major obstacle to universal acceptance of the steam-and-gas combined cycle is its fuel dependency on clean gaseous or liquid fuels.

In this combined cycle, the turbine condensate is mixed with sufficient LP economizer flow in the deaerator to raise the temperature sufficiently to avoid corrosion at the cold end of the heat-recovery steam generator. The use of regenerative feedwater heaters is avoided because of the availability of excess heat in the stack for that purpose. Part-load heat rate can be maintained at greatly reduced load in the multigas turbine plants because the units can be put in service sequentially. A so-called hockey-stick curve of part-load performance is shown in Fig. 6-16 for a combined-cycle plant utilizing four gas turbines. Below about 75% load, a gas turbine should be removed from service for best plant efficiency.

6.3.10 Noncondensing-Turbine Efficiencies

The straight noncondensing turbine generator is widely used in the process industries. The flow through a noncondensing turbine is dependent on the heat required in the process, and in order to determine the heat leaving the turbine, it is also necessary to know the enthalpy of the exhaust steam. Figure 6-17 is a plot of the mechanical and electrical efficiency of several turbine generators from 20% to 100% load. This curve permits the derivation of the internal wheel efficiency (η_{evp}) of the turbine, and ultimately the exhaust enthalpy.

6.3.11 Automatic-Extraction-Turbine Efficiencies

The automatic-extraction turbine provides the capability of delivering extraction steam at more than one process pressure simultaneously. When a condensing element is used, the kilowatt output of the unit can be maintained if the process flow varies, and will permit generation in excess of by-product power capability. The base efficiency for an automatic-extraction turbine is less than that of a straight condensing or straight noncondensing turbine because of (1) the introduction of a second control stage and accompanying parasitic losses, (2) the partial-load loss resulting when the high-pressure section of the automatic-extraction unit is passing only the nonextraction flow, and (3) the decreased pressure ratio of each section of the unit.

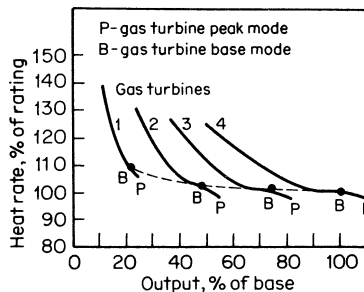


FIGURE 6-16 Output/heat rate chart for the combined-cycle power system.

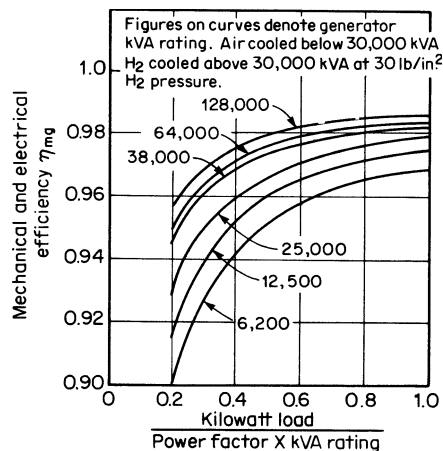


FIGURE 6-17 Typical mechanical and electrical efficiencies of turbine generators. (General Electric.)

The variation in throttle flow required for a change in extraction flow while maintaining constant kilowatt load can be derived from the available energy and efficiency of each section of the turbine. The extraction factor is a term which defines the relationship and represents the Δ throttle flow (ΔF_T) required for a Δ extraction flow (ΔF_x) of 1 lb/h at constant kilowatt load. The term represents a comparison of the used energy in the turbine below the automatic-extraction point to the total used energy of the turbine. The approximate extraction factor varies with the ratio of the theoretical steam rates for the turbine.

As extraction flow increases, the HP section of the unit generates more of the kilowatts and the LP section generates less until the steam flow to the LP stages is at the minimum necessary for cooling purposes. At this point, the maximum extraction for the load in question has been reached and further extraction flow must be accompanied at increasing output. The minimum cooling steam required varies with turbine size, extraction pressure, and the exhaust pressure. As an approximation, the minimum section flow in pounds per hour can be considered equal to the rating of the turbine in kilowatts.

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6.4 GAS TURBINES

6.4.1 Cycles

Internal combustion engines, such as conventional automotive engines, operate on the Otto cycle; injection engines operate on the Diesel cycle; and the gas or combustion turbine operates on the Brayton cycle (Fig. 6-18), also called the *gas-turbine simple cycle*. Referring to Fig. 6-19a, an axial or centrifugal compressor delivers the compressed air to the combustion system, and fuel is burned

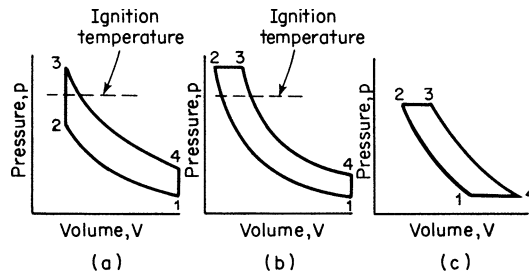


FIGURE 6-18 Ideal indicator cards (pressure-volume diagrams) for internal combustion cycles; (a) Otto cycle; (b) diesel cycle; (c) Brayton cycle. In general, phase 1-2 represents isentropic compression; phase 2-3, heat addition at constant pressure or volume; phase 3-4, isentropic expansion; and phase 4-1, heat rejection at constant pressure or volume.

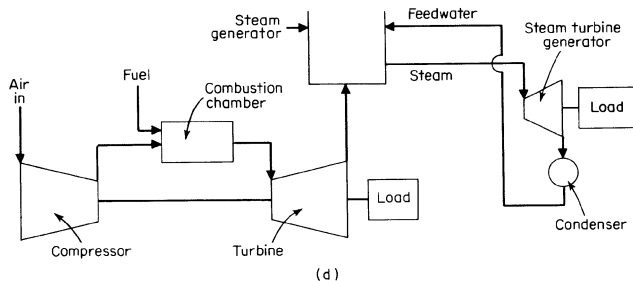
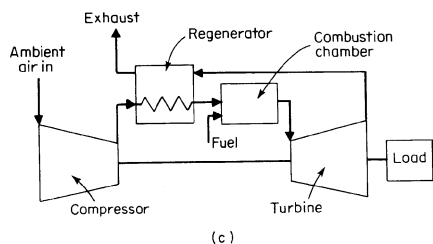
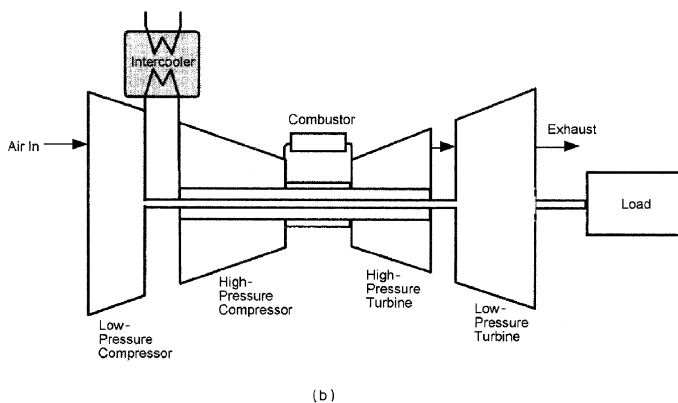
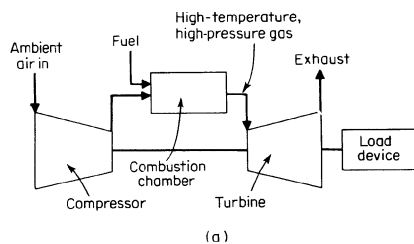


FIGURE 6-19 Typical gas-turbine cycles; (a) open, (b) intercooled (c) regenerative (d) combined.

to increase the fluid temperature. The products of combustion expand through the turbine, producing sufficient power to drive the compressor and the load. Some compressed air typically bypasses the combustor and is used to cool the turbine parts. The highest-pressure turbine airfoils contain internal cooling passages in order to maintain the metal temperature at acceptable levels for durability, while the gas path temperature is considerably higher than the metal temperature, to achieve high power and efficiency.

An improvement in power output and efficiency can be obtained through the use of an intercooler (Fig. 6-19*b*), in which air is cooled after part of the compression process. The intercooler reduces the work of compression of the high-pressure compressor and allows higher airflow and overall pressure ratio to be attained, while reducing the temperature of the cooling air for the turbine section.

Another efficiency improvement can be obtained from a regenerator or recuperator (Fig. 6-19*c*), which exchanges heat from the exhaust to the combustor inlet to reduce the fuel required to heat the gas. The highest efficiencies are available from combined cycles, where the gas turbine exhaust heat produces steam to drive a steam-turbine generator (Fig. 6-19*d*). The steam turbine output is obtained with no additional fuel input.

The most effective utilization of the fuel input is available through *cogeneration*, or combined heat and power. A typical cycle has the gas-turbine generating power and producing steam from its exhaust heat. The steam is sent to an industrial process, in some cases after generating some power in a noncondensing steam turbine. When credit is taken for the heat sent to process plus the power generated, efficiencies exceeding 80% are commonly achieved.

Simple cycles and combined cycles are by far the most commonly employed gas-turbine cycles. Some regenerative cycles were developed in the 1960s and early 1970s, but durability problems with the regenerators prevented further use. Newer regenerative cycle development started in the mid-1990s. Intercooled cycles are being studied principally as possible derivatives of commercial aircraft engines.

6.4.2 Design

Most gas turbines with outputs above 1 to 2 MW have multistage axial-flow compressors and turbines. Lower power units tend to have single-stage centrifugal compressors and radial-inflow turbines to minimize weight, size, and cost. Thermal efficiency improves with larger size, as friction and tip leakages become a smaller percentage of power produced, and multistage axial flow components become more efficient than radial stages. The same major components are included in an aeroderivative gas turbine, that is, one which was derived from an aircraft engine. This aeroderivative has two compressor sections, each driven by a separate turbine. This section of the gas turbine, called the *gas generator*, is derived from the aircraft engine. Hot gas exiting the gas generator drives the power turbine, which is connected to the load. Typical aeroderivatives differ from frame-type gas turbines in being lighter in weight and having a higher pressure ratio. Their outputs are limited to a maximum of about 50 MW because of the size of the aircraft engines from which they are derived. The largest frame-type gas turbines have outputs over 200 MW. Aeroderivatives tend to produce higher simple cycle efficiencies, whereas frame types produce the highest combined cycle efficiencies. For a given output, frame-type gas turbines tend to be somewhat less expensive than aeroderivatives.

For applications where the output shaft speed varies, as in a compressor drive, a separate power turbine is needed. This requires a multiple-shaft gas turbine (Fig. 6-20). A single-shaft gas turbine has a typical effective operating range of 85% to 105% of rated speed. With a separate power turbine to drive the load, the output shaft speed range is typically about 50% to 105% of rated speed.

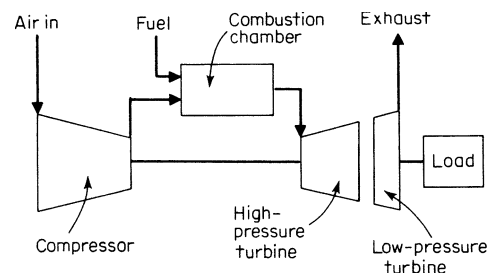


FIGURE 6-20 Multiple-shaft turbine.

6.4.3 Performance

Component efficiencies, airflow, pressure ratio, and turbine inlet (or firing) temperature are the major factors affecting gas-turbine output and efficiency. Typical multistage axial component efficiencies are in the 86% to 93% range. Material developments and turbine cooling techniques permit turbine rotor inlet temperatures to exceed 2500°F (1370°C) for the most advanced units.

6.4.4 Applications

The most familiar application of gas turbines has been for aircraft propulsion, where the turbine drives only the compressor and the remaining energy is used for thrust. The industrial gas-turbine industry started in the late 1950s, following the development of aircraft gas turbines. Generation of electric power, mechanical drive (principally gas or oil pipeline compression), and marine propulsion are the three applications of industrial gas turbines. Over 90% of the gas-turbine applications, as measured in megawatts of power produced, are in electric power generation. The majority of the navies of the world use gas turbines to propel most of their surface ships.

In electric power generation service, the combination of lower capital cost, shorter installation time, high efficiency, and environmental advantages compared to steam-turbine-based power plants has resulted in gas-turbine-based power plants having a major market for new power generation equipment.

Compliance with emissions regulations, particularly nitric oxide, or NO_x , is a major application consideration. In the late 1970s and the 1980s, water or steam was injected into the combustion reaction zone to decrease the flame temperature, which is the principal parameter affecting NO_x . Present technology is producing combustion system designs that feature premixed air and fuel in lean mixtures to reduce the flame temperature without any outside diluent.

Much work has been done to adapt gas turbines to coal fuel. Coal gasification, utilized to remove contaminants, is particularly attractive since the gas-turbine combined cycle can be integrated into the gasification process for improved efficiency. Air extracted from the gas turbine can be used as the source of oxidant for the gasification process, and the steam produced in the process can be expanded through the steam turbine. Figure 6-21 is a block diagram showing how various components relate in an integrated gasification combined cycle (IGCC).

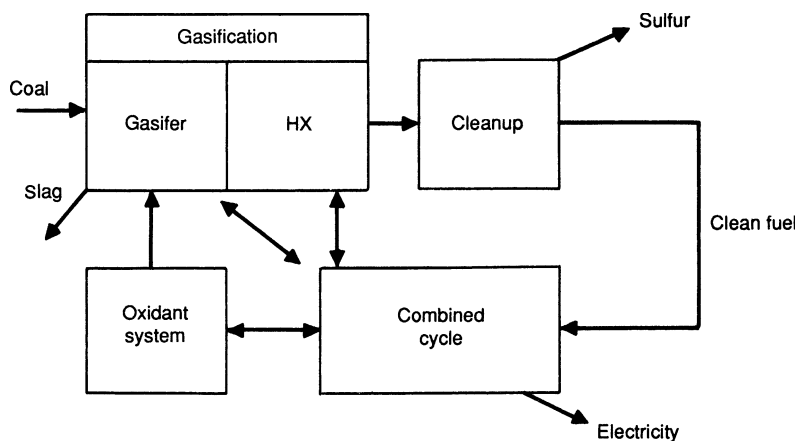


FIGURE 6-21 Possible integration among components of IGCC system.

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